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Technical Note

Effect of interphase matter transfer on condensation on low-finned tubes—a theoretical investigation

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Abstract

The paper gives theoretical results for condensation on low-finned tube in which the temperature drop at the liquid–vapour interface due to interphase matter transfer (interface resistance) is included. The condensation coefficient is taken as unity. Results show, for the case of steam, that in the regions of the fin surface where the condensate film is very thin, the local heat flux can be reduced by a factor of around 2 when the interface resistance is included. Theoretical results for the top of the tube show a significant drop in average vapour-to-surface heat-transfer coefficient with decrease in vapour pressure (around half associated with interface resistance and half due to fluid property variation) in line with earlier measurements.

Keywords: Condensation; Enhanced heat transfer; Low-finned tube; Interface matter transfer; Steam; Theoretical analysis

1. Introduction

The problem of condensation on horizontal low-finned tubes is now well understood. The fact that heat-transfer enhancement can significantly exceed area enhancement through finning is attributable to surface tension induced pressure gradients which thin the condensate film in the vicinity of strong changes in curvature of the condensate surface. Several researchers have contributed to the understanding of this problem and the earlier works are discussed by Marto [1], Honda and Rose [2] and in papers referenced in the present paper. Two models of Honda and Nozu [3] and Rose [4] embody the same principles and give results in good agreement with each other and with experimental data for various fluids. They differ in that the model of Honda and co-workers is wholly theoretical and while that of Rose used dimensional analysis backed by experimental data for three fluids and a range of tube geometries and gives algebraic results.

Of particular concern in the present paper is the influence on this problem of interphase matter transfer or "interface resistance" which was not included in the above-mentioned models. Departure from equilibrium at the vapour–condensate interface, and consequent "temperature jump" is generally unimportant in condensation problems, notable exceptions being dropwise condensation and condensation of metals. Theory shows that interphase resistance is only significant at condensation rates higher than normally found in practice. Where interface resistance is significant its effect is larger at low pressure.

Failure of the models to predict low pressure data for condensation of steam on low-finned tubes (Wanniarachchi et al. [5]), and the fact that very intense condensation can occur in the regions where the condensate film is very thin, led Briggs and Rose [6] to amend the result of Rose [4] by incorporating an interface resistance in an approximate way. Their results indicated that the low pressure steam data of Wanniarachchi et al. could indeed be explained on this basis. In the present work the interface temperature drop is included from the outset in a theoretical analysis along the line of Honda and Nozu [3], Honda et al. [7,8].

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Nomenclature

d_c	coolant side diameter, see Fig. 2
d_o	diameter at fin tip, see Fig. 2
d_r	diameter at fin root, see Fig. 2
f	condensation coefficient
g	specific force of gravity
h	fin height
h_{fg}	specific enthalpy of evaporation
$\dot{m}$	condensing mass flux across vapour–condensate interface
p	fin pitch
P	pressure
P_s	saturation pressure
q	heat flux
q_m	measured mean heat flux (whole tube) [5]
r	radius of curvature of the condensate surface in the fin cross-section, see Fig. 2
r_r	radius at fin root, see Fig. 2
r_w	radius of curvature of fin surface in the fin cross-section, see Fig. 2
R	specific ideal-gas constant for vapour
s	fin spacing at fin root, see Fig. 2
t	fin thickness, see Fig. 2
T	temperature
T_s	saturation temperature
T_w	tube wall temperature
ΔT_i	interphase temperature drop due to "interface resistance"
x, y	coordinates along and normal to fin surface, see Fig. 2
x_r	x coordinate at mid-point of interfin space, see Fig. 2
X, Y	fixed coordinates defined in Fig. 2

Greek symbols

δ	condensate film thickness measured normal to fin surface
$\varepsilon_{\Delta T}$	enhancement ratio (heat flux for finned tube divided by heat flux for smooth tube both based on smooth tube area at fin root diameter and for same ΔT
γ	ratio of principal specific heat capacities of vapour
φ	angle measured from the top of tube, see Fig. 2
θ	fin tip half angle
λ	condensate thermal conductivity
ν	kinematic viscosity
ρ	density
σ	surface tension
ψ	angle between the normal of fin surface and Y coordinate, see Fig. 2
ξ	see Eq. (1)
ζ	defined in Eq. (6)

Subscripts

cal	calculated
exp	experiment
f	fin
c	coolant
i	interface
l	condensate
p	pitch
r	fin root
s	saturation
v	vapour
w	wall
x	local

2. Interphase matter transfer

The fact that, during condensation and evaporation, there is a net transfer of molecules to or from the surface leads to a temperature difference between the surface of the liquid and that in the vapour a few mean free paths distant from the interface. Various related theoretical approaches to this problem have been proposed over the past 50 years or so. These are reviewed by Rose [9]. The theoretical predications are strongly affected by the value taken for the so-called condensation coefficient f, [1] the fraction of those vapour molecules striking the liquid surface which remain in the liquid phase. With low values of f theory indicates high interface temperature difference. Although some early measurements suggested low values (around 0.01 or lower) it is now generally thought that f is near or equal to unity. When f is taken as unity several of the more recent theoretical results, when linearised, give almost identical results and indicate, for condensation of monatomic gases, that

$$\dot{m} = \xi[P - P_s(T_s)]/\sqrt{RT_s} \tag{1}$$

with $\xi = 0.665 \pm 0.003$.

Le Fevre [10] has suggested that, for polyatomic gases, Eq. (1) may be corrected to give

$$\dot{m} = \frac{4(\gamma - 1)}{(\gamma + 1)}\xi[P - P_s(T_s)]/\sqrt{RT_s} \tag{2}$$

[1] The usual symbol for the condensation coefficient is σ but f is preferred here since σ is used for surface tension.

where γ is the ratio of the principal specific heat capacities of the vapour. With the Clausius–Clapeyron equation and $\rho_v \ll \rho_l$, Eq. (2) gives

$$\Delta T_i = \frac{q}{4\xi}\frac{(\gamma+1)}{(\gamma-1)}T_s\sqrt{RT_s}/(\rho_v h_{fg}^2) \tag{3}$$

3. Analysis including interface resistance

As noted above, the possible importance of interface resistance arises from the intense condensation flux that may occur where the condensate film is extremely thin i.e. in the vicinity of sharp changes in condensate surface curvature. As a first approximation to the distribution of heat flux and condensation flux over the surfaces of the fin and the interfin tube surface, Briggs and Rose [6] modified the model of Rose [4] by considering that all of the heat transfer took place through a fixed fraction of the total area. They found that the low pressure data of Wanniarachchii et al. [5] were quite well predicted by the modified theory when this fraction was set to 10% (see Fig. 1), while the atmospheric pressure steam data and the data for refrigerants were only marginally affected.

Fig. 2 illustrates the problem considered here with arbitrary fin profile. When interface resistance (see Eq. (3)) is included, conservation of momentum, with the assumption of pure conduction across the thin condensate film, yields the differential equation for the condensate film thickness:

$$\begin{aligned}&\frac{(\rho_l-\rho_v)g\cos\varphi}{3v_l}\frac{\partial}{\partial x}(\sin\psi\delta^3)-\frac{\sigma}{3v_l}\frac{\partial}{\partial x}\left\{\frac{\partial}{\partial x}\left(\frac{1}{r}\right)\delta^3\right\}\\&\quad+\frac{2(\rho_l-\rho_v)g}{3v_l d_o}\frac{\partial}{\partial x}(\sin\varphi\delta^3)\\&\quad=\frac{1}{(1+\zeta\lambda_l/\delta)}\frac{\lambda_l(T_s-T_w)}{h_{fg}\delta}\end{aligned} \tag{4}$$

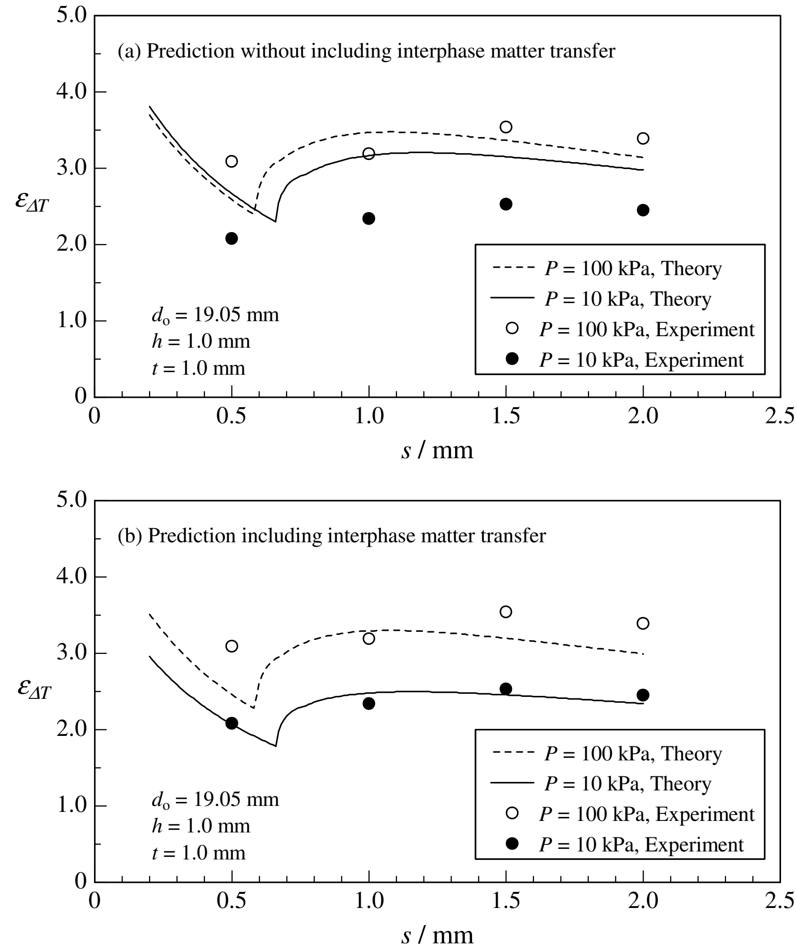

Fig. 1. Comparison of theory of Rose [4] without and with including interphase matter transfer with data of Wanniarachchi et al. [5] for steam (see Briggs and Rose [6]).

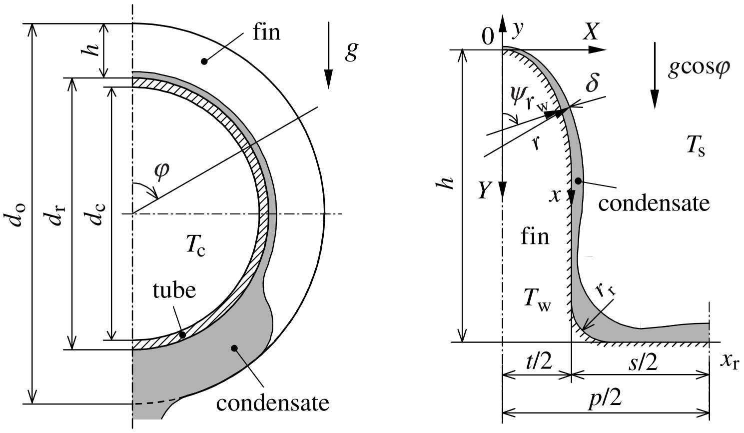

Fig. 2. Physical model and coordinates.

where

$$\frac{1}{r}=\frac{\partial^2 Y_1/\partial X_1^2}{[1+(\partial Y_1/\partial X_1)^2]^{3/2}}$$
$$=\frac{\frac{1}{r_w}+\left(\frac{2}{r_w^2}+\frac{\delta}{r_w^3}\right)\delta+\left(\frac{2}{r_w}\frac{\partial\delta}{\partial x}-\frac{\delta}{r_w^2}\frac{dr_w}{dx}\right)\frac{\partial\delta}{\partial x}-\left(1+\frac{\delta}{r_w}\right)\frac{\partial^2\delta}{\partial x^2}}{\left[\left(1+\frac{\delta}{r_w}\right)^2+\left(\frac{\partial\delta}{\partial x}\right)^2\right]^{3/2}} \tag{5}$$

$$\zeta=\frac{1}{4\xi}\frac{(\gamma+1)}{(\gamma-1)}T_s\sqrt{RT_s}/(\rho_v h_{fg}^2) \tag{6}$$

The boundary conditions are

$$\partial\delta/\partial\varphi=0 \quad \text{at } \varphi=0 \tag{7}$$

$$\partial\delta/\partial x=\partial^3\delta/\partial x^3=0 \quad \text{at } x=0 \text{ and } x_r \tag{8}$$

For the present purpose of assessing the possible significance of interphase matter transfer resistance we consider only the uppermost part of the tube ($\varphi=0$). Moreover we shall only consider the case of a high-conductivity tube material so that the temperature along the fin surface is uniform. In the earlier models [3,4] the outer surface temperature of the condensate film was taken to be uniform and equal to the vapour saturation temperature. Here we include the local temperature drop in the vapour at the interface given by Eq. (3) so that the condensate surface temperature is no longer uniform and depends on the local heat flux. The local heat flux is given by

$$q_x=\frac{1}{(1+\zeta\lambda_1/\delta_x)}\frac{\lambda_1(T_s-T_w)}{\delta_x} \tag{9}$$

The average heat flux q_p along x from 0 to x_r is defined on the projected area basis as

$$q_p=\frac{2}{p}\int_0^{x_r}q_x\,dx \tag{10}$$

Eq. (4), subject to boundary conditions (7) and (8), was solved numerically by a finite difference scheme described in Honda et al. [8].

4. Results

For purposes of comparison, the best performing tube in the measurements of Wanniarachchi et al. [5] was chosen, i.e. spacing between fins 1.5 mm. These data have been carefully measured from heat-transfer coefficient versus heat flux plots and represented in Fig. 3 as heat flux versus temperature difference. A clear effect of pressure may be seen, the vapour-to-surface heat-transfer coefficients being around 30% smaller at the lower pressure. Fig. 4 shows an example of the calculated condensate film profile along the fin surface for the upper surface of the tube ($\varphi=0$). The condensate film is seen to be extremely thin at the corner of the fin tip and

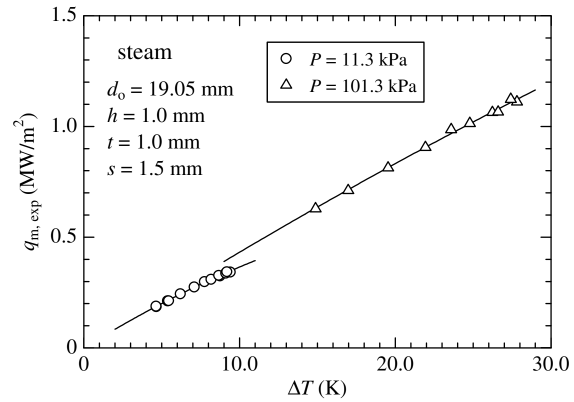

Fig. 3. Dependence of measured heat flux on vapour-to-surface temperature difference [5].

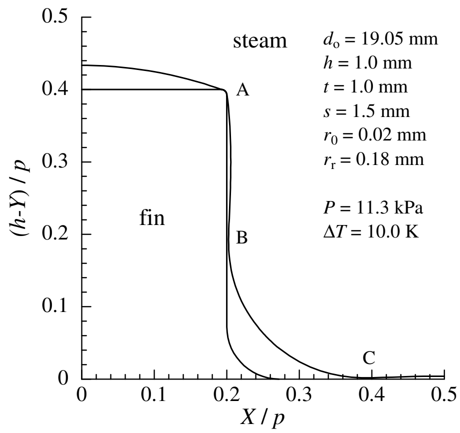

Fig. 4. Profiles of fin and calculated condensate film including interface resistance ($\varphi = 0$).

at locations towards the fin root on the fin flank and interfin tube surface (see locations A, B, C in Fig. 4). Fig. 5 shows examples of calculated profiles of heat flux (at $\varphi = 0$) along the surface from the mid-point of the top of the fin along the flank to the mid-point of the interfin tube surface. Results are shown for cases without and with inclusion of the interface resistance. Twin peaks may be seen on either side of the corner of the tip (where abrupt changes in surface curvature occur) and further smaller peaks lower on the fin flank and on the interfin tube surface. The difference between the results without and with interface resistance (masked by the logarithmic scale) is larger where the heat flux is higher as expected. The peak heat fluxes at location A for the two cases are 12.3 and 7.0 MW/m^2.

Fig. 6 shows calculated results for the average heat flux along the surface at the top of the tube ($\varphi = 0$) over the range of ΔT obtained by Wanniarachchi et al. [5]. Also included are the measurements of Wanniarachchi et al. for the average heat flux for the whole tube. The calculated results are shown both excluding and including interface resistance. It is seen that the effect of interface resistance is stronger at the lower pressure (due to low vapour density, see Eq. (3)). As expected the calculated heat fluxes for the top of the tube ($\varphi = 0$) are significantly higher than the measured average values for the whole tube perimeter. It may be seen that the calculated drop in heat flux with pressure is attributable roughly in equal proportion to variable properties (i.e. when interface resistance is not included) and interface resistance. The ratio of the calculated mean heat flux at atmospheric pressure to that at the lower pressure is 1.34 when the interface resistance is included and 1.18 when interface resistance is omitted from the calculation. The ratio of the measured values for the whole tube is 1.30. (Note—for the purpose of obtaining these ratios the experimental and theoretical curves were fitted by the equation $q = a\Delta T^{3/4}$ so that the ratio of heat fluxes or heat-transfer coefficient (ratio of a values) is independent of ΔT. The lines shown in the figure for the experiment results are from these curve fits.) As expected, the calculated (including interface resistance) ratio for the top of the tube ($\varphi = 0$), where the heat flux is highest, exceeds the ratio obtained from the measured average values for the whole tube. When interface resistance is omitted the calculated ratio for the top of the tubes is smaller than the value found from the measured average results.

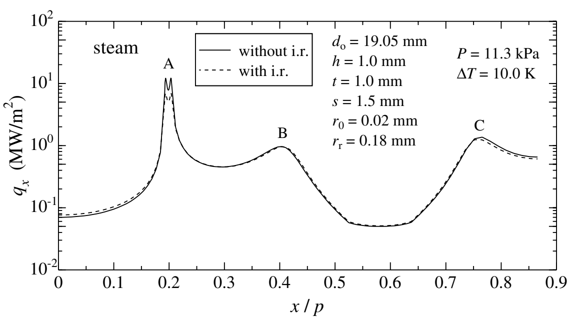

Fig. 5. Calculated profiles of heat flux along fin surface without and with interface resistance ($\varphi = 0$, i.r. denotes interface resistance).

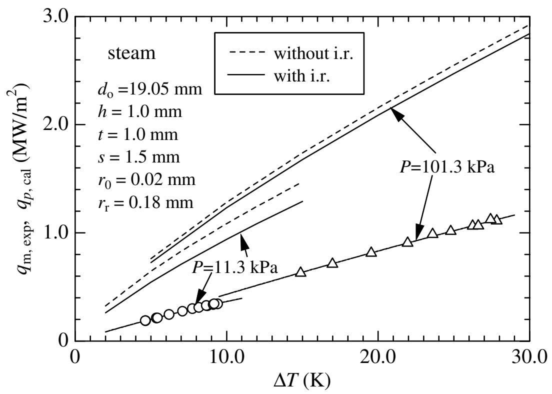

Fig. 6. Dependence of measured and calculated heat flux ($\varphi = 0$) against vapour-to-surface temperature difference.

5. Conclusion

The present results are in general accord with experimental data and earlier approximate calculations [6] and support the conclusion that interphase matter transfer (interface resistance) has significant effect on heat transfer for condensation of steam on low-finned tubes and should be taken account of in calculations, particularly for low vapour pressure. It is to be noted that a condensation coefficient of unity has been assumed in the present work; lower values would lead to a stronger effect i.e. lower heat transfer.

References

[1] P.J. Marto, An evaluation of film condensation on horizontal integral-finned tubes, Trans. ASME J. Heat Transfer 110 (1988) 1287–1305.

[2] H. Honda, J.W. Rose, Augmentation techniques in external condensation, in: Handbook of Phase Change: Boiling and Condensation, Taylor Francis, 1999, pp. 605–620.

[3] H. Honda, S. Nozu, A prediction method for heat transfer during film condensation on horizontal low integral fin tubes, Trans. ASME J. Heat Transfer 109 (1987) 218–225.

[4] J.W. Rose, An approximate equation for the vapour-side heat-transfer coefficient for condensation on low-finned tubes, Int. J. Heat Mass Transfer 37 (1994) 865–875.

[5] A.S. Wanniarachchi, P.J. Marto, J.W. Rose, Film condensation of steam on horizontal finned tubes: Effect of fin spacing, Trans. ASME J. Heat transfer 108 (1986) 93–99.

[6] A. Briggs, J.W. Rose, Effects of interphase matter transfer and non uniform wall temperature on a model for condensation on low-finned tubes, Int. J. Transport Phenomena 1 (1998) 41–49.

[7] H. Honda, H. Takamatsu, O. Makishi, Numerical analysis of film condensation on a horizontal two-dimensional fin tube, in: Proc. Eurotherm Seminar 47, Heat Transfer in Condensation, Paris, 4–5 October 1995, Elsevier, 1996, pp. 48–54.

[8] H. Honda, H.S. Wang, S. Nozu, A theoretical study of film condensation in horizontal microfin tubes, Trans. ASME J. Heat Transfer 124 (1) (2002) 94–101.

[9] J.W. Rose, Interphase matter transfer, the condensation coefficient and dropwise condensation, in: Proc. 11th Int. Heat Transfer Coeff., Kyongju, Korea, vol. 1, 1998, pp. 89–104.

[10] E.J. Le Fevre, Private communication, (a) An estimate of bulk viscosity in ideal gases, (b) Dissipative normal stress in vapour near liquid–vapour interface, 1964, See Appendix G of Ref. [11].

[11] J.W. Rose, Dropwise condensation of steam on vertical planes, Ph.D. thesis, Univ. London, 1964.